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(71) Applicant: SCANIA CV AKTIEBOLAG [SE/SE]; S-151 87 Södertälje (SE).

(72) Inventor: SLAPAK, Dieter; Förvaltarvägen 20, S-151 47 Södertälje (SE).

(74) Agent: WALDEBÄCK, Hans; Scania CV AB, Patents, S-151 87 Södertälje (SE).

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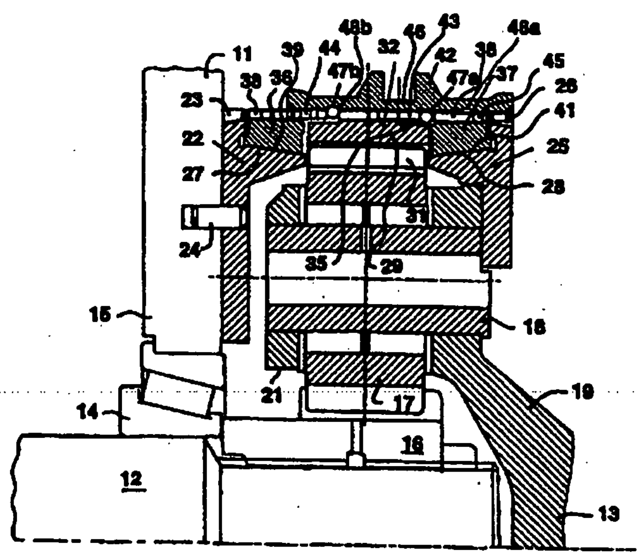

(57) Abstract

Planetary gearing comprising a sun gear (16), planet gears (17) disposed around the sun gear and rotatably connected to an output shaft (13), and a ring gear (29) disposed around the planet gears (17). A clutch ring (22, 25) and a synchronising ring (36, 37) are located one on each side of the ring gear (29) and a coupling sleeve (42) is arranged externally of the clutch rings (22, 25), synchronising rings (36, 37), and ring gear (29), and is displaceable to couple the ring gear (29) with either of the clutch rings (22, 25). The coupling sleeve (42) carries a pair of resilient clips (47) which are abuttable against the synchronising rings (36, 37), and which are resiliently deformable to allow the sleeve (42) to be displaced relative to the clips (47) for fully engaging the clutch rings (22, 25) after synchronisation.

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Planetary Gearing

Field of Invention

This invention relates to planetary gearing according to the preamble to attached claim 1. Such a planetary gear is used in particular but not exclusively for auxiliary gear boxes intended in use to be inserted between a main gear box and a transmission to the driving wheels of a motor vehicle, especially heavy duty trucks.

Background of Invention

Auxiliary gear boxes are typically used for doubling the number of gear change possibilities and usually comprise planetary gearing by means of which the gear change possibilities of the vehicle can be diverted into a low gear range and a high gear range. In the high gear range there is no gear reduction and in the low gear range use is made of the gear changing in the planetary gearing system.

One known embodiment of a planetary gearing is disclosed in US-A-5083993 and discloses gearing having an axially displaceable coupling sleeve which can be coupled with clutch rings located one on either side of a ring gear. Synchronising rings are also located one on each side of the planet gears, between the planet gears and a respective clutch ring. The coupling sleeve at its axial ends has recesses for accommodating spring loaded locking bodies which butt against shoulders on the synchronising rings in order to transmit axial load to the synchronising rings. The accommodating of the locking bodies and their springs within the coupling sleeve provides for a complicated synchronising arrangement with many parts.

The present invention seeks to provide a synchronising arrangement for a planetary gearing which is simple and has a minimal number of parts.

Statement of Invention

Accordingly there is provided a planetary gearing according to the characterising part of claim 1. The dependent claims relates to preferred embodiments of the invention. Preferably the resilient means is coaxial circlip, which in a first condition is in a relaxed state and in a second condition is in a more stressed state.

Preferably the circlip is held in an internal groove in the coupling sleeve and when subject to an axial load contracts in circumference and is dislodged from groove, the circlip being slidable over the internal surface of the coupling sleeve.

Conveniently a respective resilient means is provided for abutment with each synchronising ring and preferably in one axial direction of movement of the coupling sleeve one of said circlips abuts one side face of the ring gear external teeth and the second of said circlips abuts the synchronising ring on other side of the ring gear, and in the opposite axial direction of movement of the coupling sleeve the second clip abuts said other side face of the ring gear external teeth and said one circlip abuts the synchronising ring on said one side of the ring gear.

Preferably the internal teeth on the coupling sleeve comprise three sets of teeth, a first set located in an axially central portion of the sleeve, with the second and third set of teeth being located at each end portion of the sleeve and being spaced from the first set of teeth, the axial gaps between first set of teeth and the second and third sets of teeth accommodating the relative displacement of the respective circlips.

Description of Drawings

The invention will be described by way of example and with reference to the accompanying drawings in which

Fig. 1 is a longitudinal section through planetary gearing according to the invention showing only a portion above the longitudinal axis of rotation,

Fig. 2 is a similar drawing to that of Fig. 1 showing the gearing in a high range condition,

Fig. 3 is an intermediate step going from a low range to a high range condition, showing engagement of a synchronisation cone,

Fig. 4 is an intermediate step going from a low range to a high range condition showing the end of the synchronising process, and

Fig. 5 is a sectional view on the line V-IV in Fig. 1.

Detailed Description of Invention

With reference to Fig. 1, Fig. 2, and Fig. 5 of the drawings there is shown a part section through an auxiliary gear box for fitting to the main gear box of a heavy vehicle such as a

truck or bus.

The auxiliary gearbox comprises a housing 11 having an input shaft 12 passing therein from the main gearbox not shown. The input shaft 12 is rotatably mounted in the main gear box and housing 11 by bearings 14 mounted in an inner end wall 15 of the housing 11.

A sun gear 16 is mounted rotatably fast with the input shaft 12 by means of splines. The sun gear 16 is formed with external teeth which mesh with at least one, and preferably a plurality of, for example five, circumferentially spaced planet gears 17. Each planet gear 17 is rotatably mounted on a tubular stub shaft 18 one end of which is secured to a planet gear carrier 19 and to the other of which is secured to a planet gear keeper 21. The planet gear keeper 21 retains the respective planet gear on its respective stub shaft 18. The planet gear carrier 19 is integral with an output shaft 13 which is rotatably mounted in an outer end wall not shown on the gear box housing 11 coaxially with the input shaft, as is well known in the art, typically by means of bearings as shown for the input shaft. Rotation of the planet gears 17 by the sun gear 16 causes the planet carrier 19 to rotate.

A first clutch ring 22 formed with external axially extending teeth 23 is mounted coaxially of the shaft 12 rotationally fast on the gearbox housing inner end wall 15 by means of pins 24, to one side of the planet gears.

A second clutch ring 25 formed with external axially extending teeth 26 is coaxially mounted rotationally fast on the planet gear carrier 19 to which it is secured by welding but could also be held fast by means of teeth or splines on its radially inner peripheral margin engaging with like teeth or splines on the planet gear carrier 19. The second clutch ring 25 is located on the other side of the planet gears 17 from the first clutch ring 22. The two clutch rings 22, 25 have on their sides adjacent the planet gears 17 radially outwardly directed frustoconical surfaces 27, 28.

A coaxial gear ring 29 is located axially between the two clutch rings 22, 25 and concentrically with the planet gears 17. The gear ring 29 has internal teeth 31 for meshing with the planet gears 17, and has external teeth 32 which are in radial alignment with and have the same form as the external teeth 23, 26 on the two clutch rings 22, 25. The coaxial

ring gear 29 is located in axial alignment with the planet gear 17 between a pair of synchronising rings 36, 37. The external teeth 32 are located axially centrally of the ring gear 29 which is then formed on each axial side thereof with a lesser diameter shoulder 34, 35.

The pair of synchronising rings 36, 37 are arranged with a first synchronising ring 36 located axially between the first clutch ring 22 and the ring gear 29, and a second synchronising rin 37 located axially between ring gear 29 and the second clutch ring 25. Each synchronising ring 36, 37 is formed with external teeth 38 which are radially aligned with the teeth 23, 26 on the clutch rings 22, 25 and the external teeth 32 on the ring gear 29. Each synchronisin ring 36, 37 has a radially inwardly directed frustoconical surface 39, 41 for engagement with the like surfaces 27, 28 on the clutch rings 22, 25 so that the two synchronising rings 36, 37 are arranged in a mirror image of each other.

The axially outer ends of the external teeth 38 on the synchronising rings 36, 37 are bevelled to give a lead in to the teeth, and the axially inner ends of the teeth 38 have substantially flat side faces.

Externally of the two clutch rings 22, 25, synchronising ring 36, 37 and the ring gear 29 is located an axially displaceable coupling sleeve 42. The coupling sleeve 42 has internal teeth divided into three sets, a central set 43 for meshing with the external teeth 32 on the ring gear 29 and a respective set 44, 45 at each axial end portion of the coupling sleeve for meshing with the respective synchronising ring 36, 37 and clutch rings 22, 25. The sleeve 42 has an annular slot 46 on its outer surface in which a control fork not shown engages for displacement of the sleeve 42 axially during gear changing.

Adjacent the two end groups of teeth 44, 45, there is located axially inwardly thereof at each end an arcuate section annular groove 48 facing radially inwardly. Each groove 48 can partially accommodate one of a pair of circular cross-section circlips 47, which are located internally of the coupling sleeve 42 and coaxially thereof, one in each axial gap between the central set of teeth 43 and the two end sets of teeth 44, 45. The circlip is a resilient split ring which when located in a groove 48 is in a more relaxed condition, and when contracted to a smaller circumference then an annular groove 48 is in a more highly stressed condition.

The planetary gearing described above functions is described below. In figure 1 of the accompanying drawings the inner teeth 31 of the ring gear 29 engage with the planet gears 17. The outer teeth 32 of the ring gear 29 engage with the central teeth 43 of the coupling sleeve 42. The coupling sleeve 42 is connected to the first clutch ring 22 through the end group of teeth 44 which mesh with the external teeth 23 on the first clutch ring 22. The sleeve 42 is held stationary relative to the housing 11 and the planet gear carrier 19 is free to rotate relative to the input shaft 12 with the low gear range being operational.

Engaging the high gear range is effected by moving the sleeve 42 rightwards causing the end group 44 of teeth on the sleeve 42 to disengage from the first clutch ring 22.

Simultaneously the two circlips 47 are carried rightwards by the sleeve 42, the circlip 47a held in its groove 48a, and the other circlip 47b held in its location by friction generated by its resilient expansion within the sleeve. The two rings are displaced together until the circlip 47a abuts the adjacent teeth 38 on the second synchronising ring 37 and the circlip 47b abuts the adjacent side of the teeth 32 on the ring gear 29.

An increase in rightward bias load causes the sleeve 42 to exert an axial force on the second synchronising ring 37 through the circlip 47a which results in its frustoconical surface 41 contacting the frustoconical friction surface 28 on the second clutch ring 26. This is the condition shown in Fig. 3.

As the increased rightwards load is applied to the sleeve 42, the circlip 47a is caused to contract and snap out of its groove 48a and the sleeve 42 slides rightwards. The increased load on the second synchronising ring 37 fully engages the synchronising cone surfaces 41,28. This is the condition shown in Fig. 4.

The second synchronising ring is therefore subject to an accelerating effect which increases the rotational speed of the coupling sleeve 42 until there is substantially no difference in rotational speeds between the sleeve 42 and the synchronising ring 37 and second clutch ring 25. When the rotational speed has been equalised, the axial force which the adjacent end set of teeth 45 on the sleeve 42 exerts on the bevelled surfaces of the second clutch ring teeth 26 is sufficient to turn the sleeve 42 to a position in which the adjacent end group of teeth 45

can slide into the gaps between the adjacent teeth 26 on the second clutch ring 25.

The sleeve 42 is now displaced rightwards to fully engage the end group of teeth 45 with the second clutch ring 25. The two circlips 47a, 47b thus slide axially within the gaps between the sets of teeth, so that the circlip 47b returns to its groove 48b. This is the condition shown in Fig. 2, in which the sleeve 42 is connected to the second clutch ring 25 through the end group of teeth 45 which mesh with the external teeth 26 on the second clutch ring 25. Since the ring gear 29 is now fixed relative to the output shaft 13, the gearing is operating in a high range in which the output shaft 13 will rotate at the same speed as the input shaft 12. In this condition the right hand circlip 47a is located axially between the external teeth 38 of the second synchronising ring 37 and the internal central set of teeth 43 on the coupling sleeve. The circlip 47a has been displaced from its adjacent groove 48a and is in a contracted stressed condition. The left hand circlip 47b is accommodated in its respective groove 48b and is in relaxed condition but adjacent the teeth 32 of the ring gear 29.

When the low gear range of the planetary gearing is engaging the control fork not shown operates to move the coupling sleeve 42 to the left displacing the coupling sleeve 42 from the condition shown in figure 2 to the condition shown in figure 1. When the gear range change takes place under operating conditions the coupling sleeve 42 displaces the two circlips 47 simultaneously, and the end set of teeth 45 on the sleeve 42 disengages from the external teeth 26 of the second clutch ring 25 and thereby disengages from the output shaft 13.

The sleeve 42 continues its leftward movement carrying the two circlips with it, the circlip 47b held in the groove 48b and circlip 47a held in position by friction between itself and the inner surface of the sleeve generated by its resilient bias. Simultaneously, the left hand circlip 47b abuts the teeth 38 on the first synchronising ring 36 and the circlip 47a abuts the external teeth 32 on the ring gear 29.

An increase in the leftwards bias causes the sleeve 42 to exert an axial force on the first synchronising ring 46 which results in its conical surface 39 contacting the conical surface 34 on the stationary first clutch ring 22. An increased leftwards bias load causes the circlip 47b to resiliently deform and contract and snap out of its groove 48b, as the sleeve slides

leftwards. The first synchronising ring 36 is therefore subject to a braking load which slows the relative rotational speed of the ring gear 29, and sleeve 42.

After synchronous rotation has been achieved and the relative rotation has been braked completely, the axial force which the adjacent end group of teeth 44 exert on the bevelled surfaces of the first clutch ring teeth 23 is sufficient to turn the sleeve 42 to a position in which the adjacent end group of teeth 44 can glide into the gaps between the adjacent teeth 23 on the sleeve 42.

The sleeve 42 can now be displaced further leftwards to fully engage the first clutch ring. The two circlips 47 which are held axially stationary thus axially slide in their respective gaps between the sets of teeth and are displaced rightwards relative to the sleeve 42. the right hand circlip 47a returning to its groove 48a as shown in figure 1.

In another embodiment of the invention the first clutch ring could be fixed to the sun gear or input shaft and the second clutch ring fixed to the gearbox housing.

The above described gearbox has the advantage that it is axially short, and of simple construction . Furthermore, because of the snap out characteristic of the circlips from their respective grooves, there is a positive feed back to a driver operating the control fork.

Furthermore the number of teeth 38 on the synchronising rings 36, 37 can differ from the number of teeth 32 on the ring gear 29. This gives a greater number of different combinations for connections thereby reducing problems with sun gears having facetted teeth caused by having only a few combinations.

Claims

1. Planetary gearing comprising a planetary gear carrier (19), planet gears (17) mounted on the gear carrier (19), a ring gear (29) having internal teeth (31) for meshing with the planetary gears (17) and external teeth (32), a clutch ring (22,25) and synchronising ring (36,37) arranged on each side of the ring gear (29) and having radially aligned external teeth (23,26,38) which align with the external teeth (32) on the ring gear (29), and a coupling sleeve (42) arranged around and being axially displaceable relative to the ring gear (29) and said rings (22,25,36,37) and having internal teeth (43,44,45) for meshing with the said external teeth, **characterised** in that the coupling sleeve (42) carries a resilient means (47) which in a first condition serves as an abutment for transmitting an axial load to a synchronising ring (36,37) and in a second condition permits axial displacement of the coupling sleeve (42) relative to both said resilient means (47) and to said synchronising rings (37,37).

2. Planetary gearing as claimed in Claim 1, **characterised** in that the resilient means (47) in the first condition is in a more relaxed state and in the second condition is in a more stressed state.

3. Planetary gearing as claimed in Claim 1 or Claim 2, **characterised** in that the resilient means (47) comprises a coaxial circlip (47) which is held in an internal groove (48) in the coupling sleeve (42) and when subject to an axial load contracts in circumference and is dislodged from groove (48), the circlip being slidable over the internal surface of the coupling sleeve.

4. Planetary gearing as claimed in Claim 3, **characterised** in that the circlip (47) has a round radial cross section and is housed in an arcuate section groove (48) in the coupling sleeve.

5. Planetary gearing as claimed in any one of Claims 1 to 4, **characterised** in that a respective resilient means (47) is provided for abutment with each synchronising ring (36,37).

6. Planetary gearing as claimed in Claim 5, **characterised** in that in one axial direction of movement of the coupling sleeve (42) one of said circlip (47a) abuts one side face of the ring gear external teeth (32) and the second of said circlips (47b) abuts the synchronising (36) on other side of the ring gear (29), and in the opposite axial direction of movement of the coupling sleeve (42) the second circlip (47b) abuts said other side face of the ring gear external teeth (32) and said one circlip (47) a abuts the synchronising ring (37) on said one side of the ring gear (29).

7. Planetary gearing as claimed in Claim 6, **characterised** in that the internal teeth of coupling sleeve (29) comprise three sets of teeth, a first set (43) located in an axially central portion of the sleeve, with the second (44) and third (45) set of teeth being located at each end portion of the sleeve (42) and being spaced from the first set of teeth, the axial gaps between first set of teeth and the second and third sets of teeth accommodating the relative displacement of the respective circlips (47).

8. Planetary gearing as claimed in any one of Claims 1 to 7, **characterised** in that one of said clutch rings (22) is connected directly to a housing (11) surrounding the gearing, and the other clutch ring (25) is connected to one of an input shaft (12) including a sun gear (16) which meshes with the planet gears, and an output shaft (13) connected to the planet gear carrier (19).

9. Planetary gearing as claimed in any one of Claims 1 to 8, **characterised** in that the sleeve (42) has a slot (46) on its radially outer surface for accommodating a shift fork means for displacement of the sleeve.

10. An auxiliary gearbox for the main gear box of a vehicle and which doubles the number of gears available **characterised** in that the auxiliary gearbox includes planetary gearing as claimed in any one of Claims 1 to 9.

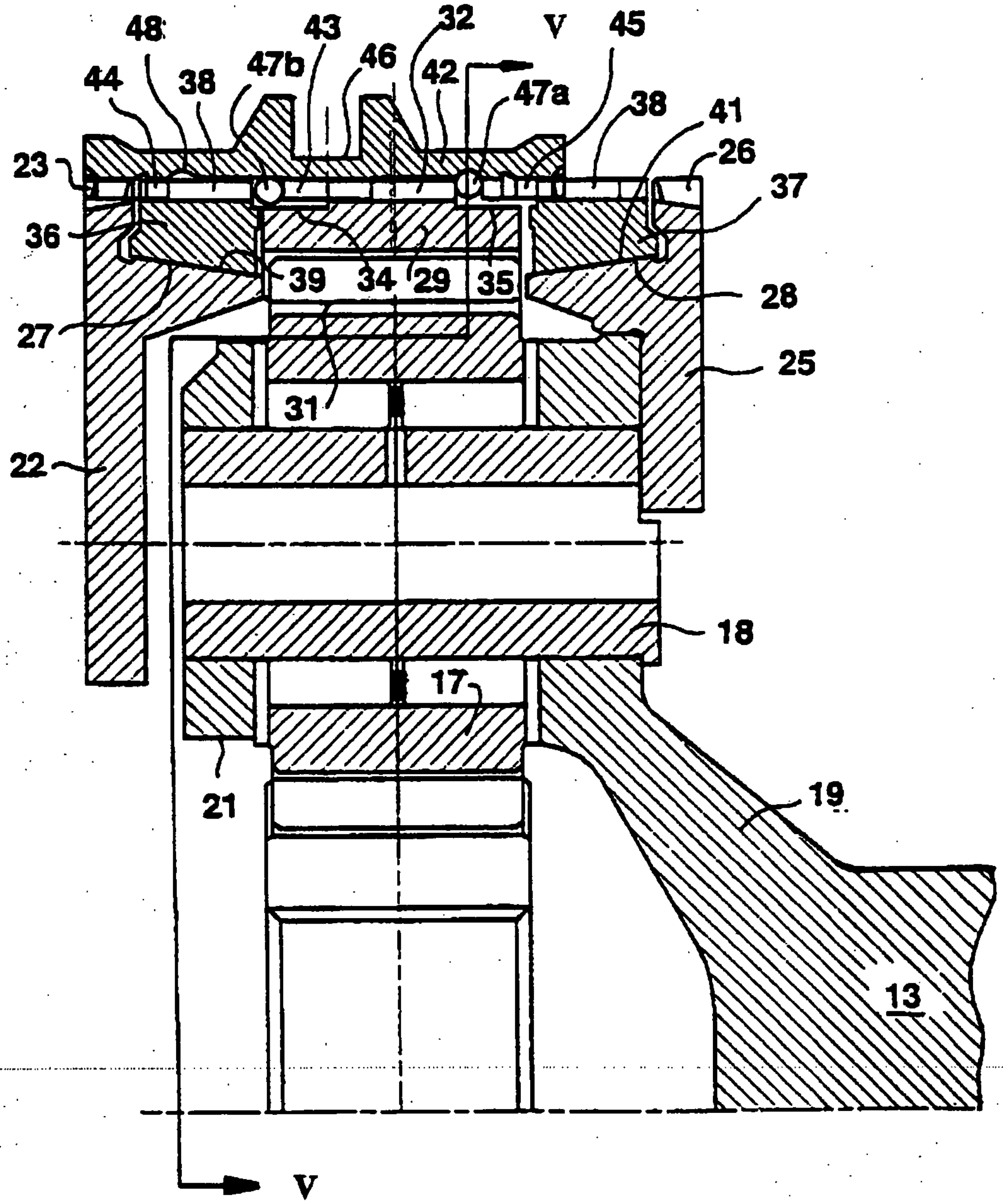
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Fig 1

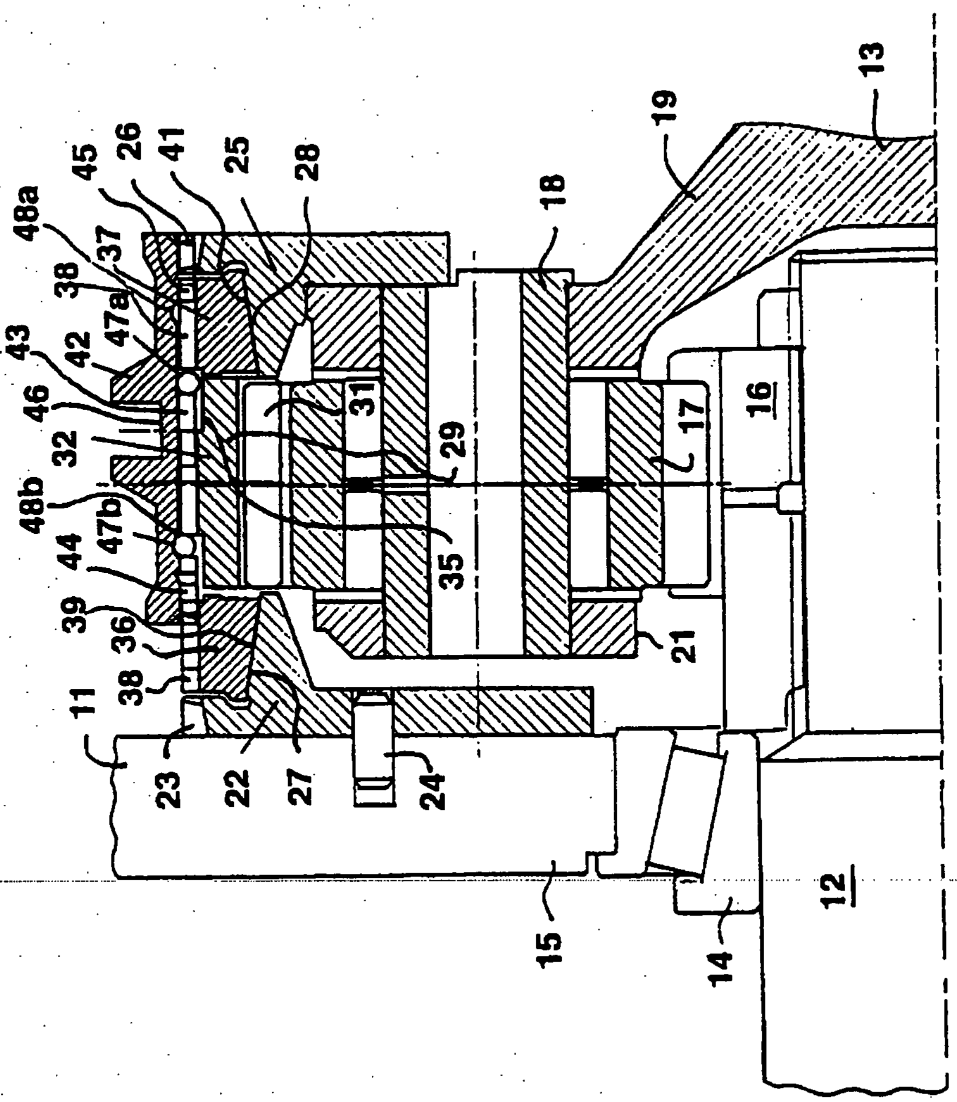
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Fig 2

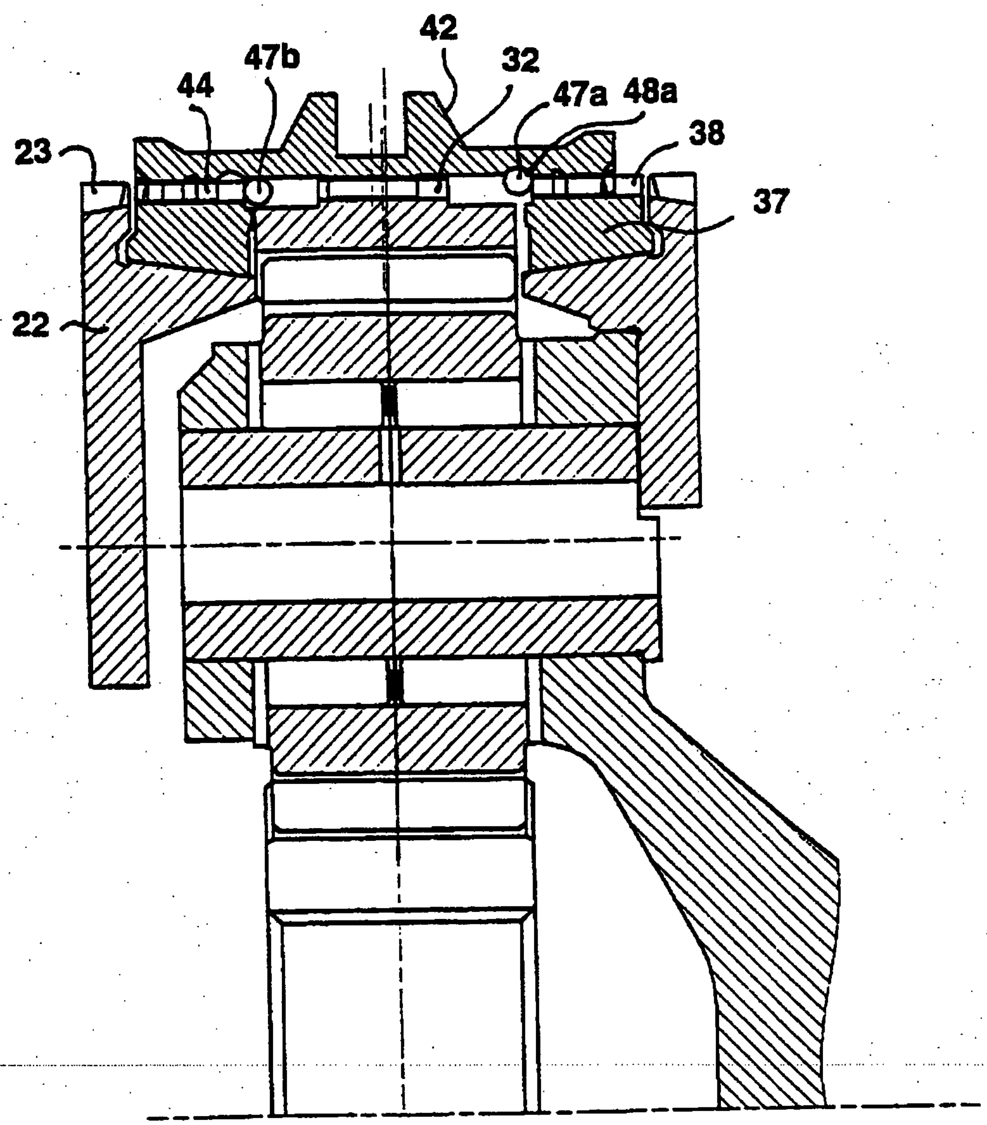

Fig 3

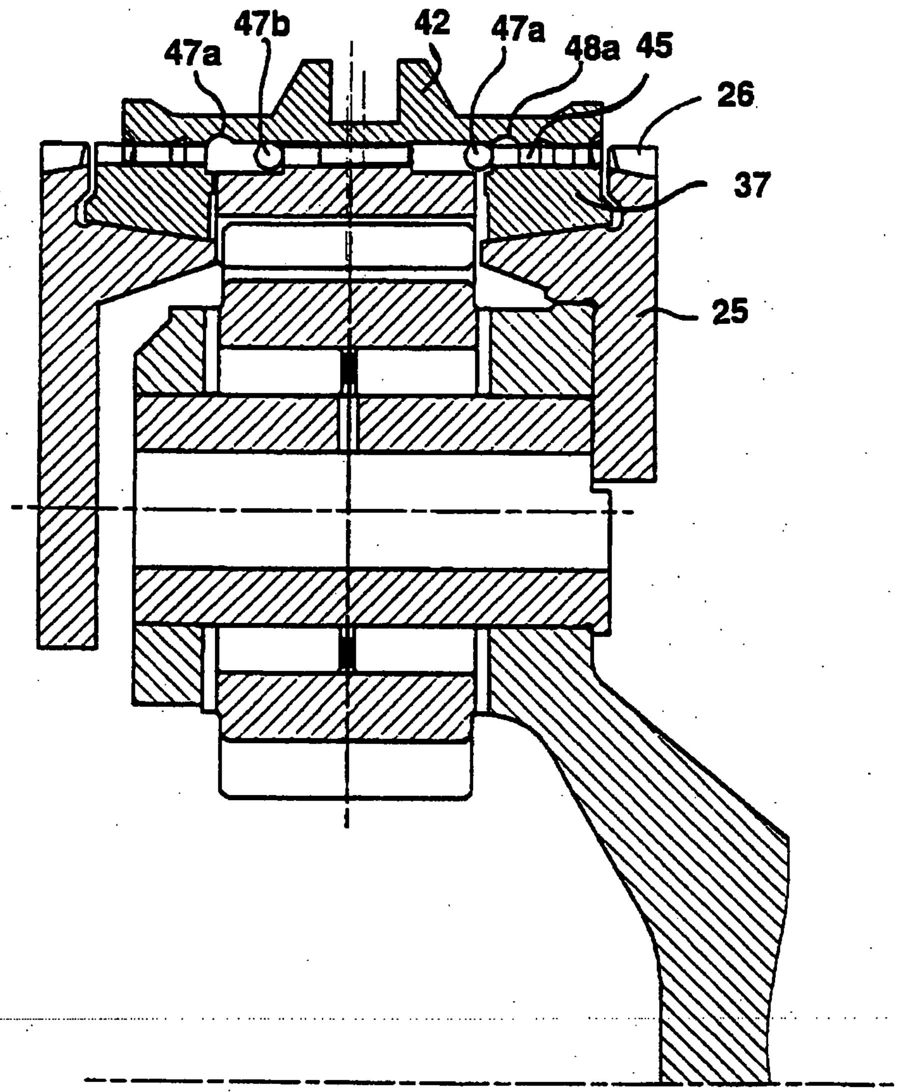

Fig4

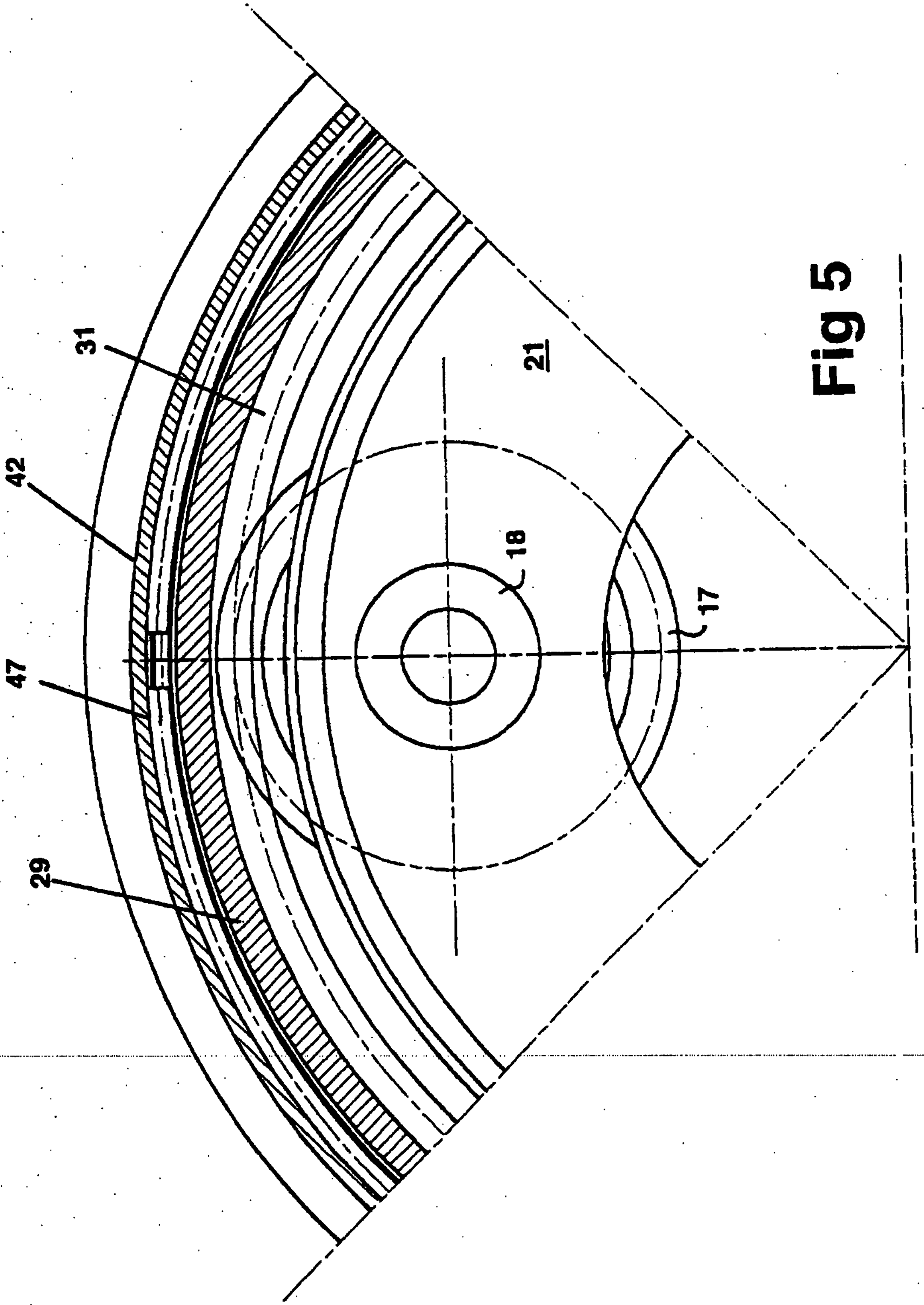

Fig 5

INTERNATIONAL SEARCH REPORT

International application No.
PCT/SE 95/01548

A. CLASSIFICATION OF SUBJECT MATTER

IPC6: F16H 3/78, F16D 23/06
According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

IPC6: F16H, F16D

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

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Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	US 5083993 A (ÖUN), 28 January 1992 (28.01.92) --	1-10
A	US 2221892 A (P. ORR), 19 November 1940 (19.11.40) --	1-10
A	DE 1750546 A (ZAHNRADFABRIK FRIEDRICHSHAFEN AG), 7 January 1971 (07.01.71) --	1-10
A	FR 1012482 A (M. CHARLES-EDOUARD HENRIOD), 10 July 1952 (10.07.52) -- --------	1-10

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6 March 1996	12.04.96
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INTERNATIONAL SEARCH REPORT
Information on patent family members

05/02/96

International application No.
PCT/SE 95/01548

Patent document cited in search report	Publication date	Patent family member(s)	Publication date
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		EP-A,B- 0423863	24/04/91
		SE-B,C- 463477	26/11/90
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US-A- 2221892	19/11/40	NONE	
DE-A- 1750546	07/01/71	NONE	
FR-A- 1012482	10/07/52	NONE	